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Momentum reversing brake for large vessels.

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MOMENTUM REVERSING BRAKE

FOR LARGE VESSELS

by

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MOMENTUM REVERSING BRAKE FOR LARGE VESSELS

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JOHN DECK, III

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ABSTRACT

This thesis covers the investigation of the performance of a scoop-like hydrodynamic brake at law Froude number flows in comparison to a flap device. The application of the brake is to large (100,000 DWT and over) seagoing vessels.

The brake was designed from a Pelton or impulse water turbine bucket. The brake derives its' force by scooping up and reversing the direction of flow of a portion of the slip stream water. The braking force thus derived should be twice that of a flap device for the same projected area perpendicular to the flow.

Two scoops in different configurations and a set of flaps were tested at low Froude number (<2) flows in the M.I.T. Propeller Tunnel. Design features and brake parameters were established in these tests. Under certain circumstances, the brake met or exceeded the predictions as to its' effectiveness.

The brake parameters were introduced into a simple math model to give an order of magnitude analysis of the application of these devices to a real vessel.

It is concluded that the use of the scoops is a significant improvement over flap devices and that further investigation is warranted particularly in the performance of flush inlet brakes and brakes discharging into a separation cavity.

Thesis Supervisor: J. Nicholas Newman
Title: Associate Professor of Naval Architecture

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I have profited from help and advice from many quarters. Those that I can name, include Professor Newman and Professor Sonin whose patience and constructive criticism helped keep this thesis on course. Professor Reed motivated the subject and gave encouragement along the way.

The Coast Guard at long last provided me with the opportunity to study. I could have never completed it without the forebearance of my wife, Susan, and my children.

Further, I wish to thank my wife, Susan, for her secretarial help and typing, in preparing this thesis.

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INTRODUCTION

BACKGROUND

In the past decade, there has been a trend in the bulk cargo trades to construct ships of 100,000 DWT displacements and up. Several tankships of 200,000 - 300,000 DWT displacement are now operating or under construction. Lloyd's Registry has released feasibility studies of a 500,000 DWT displacement tankship and has indicated a willingness to class them up to that maximum displacement. All of these gigantic tankships share, in common, a problem of long stopping times and head reach. The kinetic energy $1/2\ MV^2$ must be eliminated in order to stop the vessel. The power plants of these ships are small in comparison to their displacement and are relatively ineffective during most of the slowing down process. Sea trials of several tankers[1&2] have indicated head reaches of 12-15 ship lengths from a service speed of usually 15-16 knots. The equation of motion can be applied in this case. $F = -Ma$ where F is the braking force and M the mass of the ship and its entrained mass of water, a, of course is the deceleration of the ship system. F is composed of the hull resistance R, the propeller thrust T, and perhaps, some auxiliary breaking force B. All of these forces act parallel to the centerline of the ship. Figure 1 is a graphical representation of the

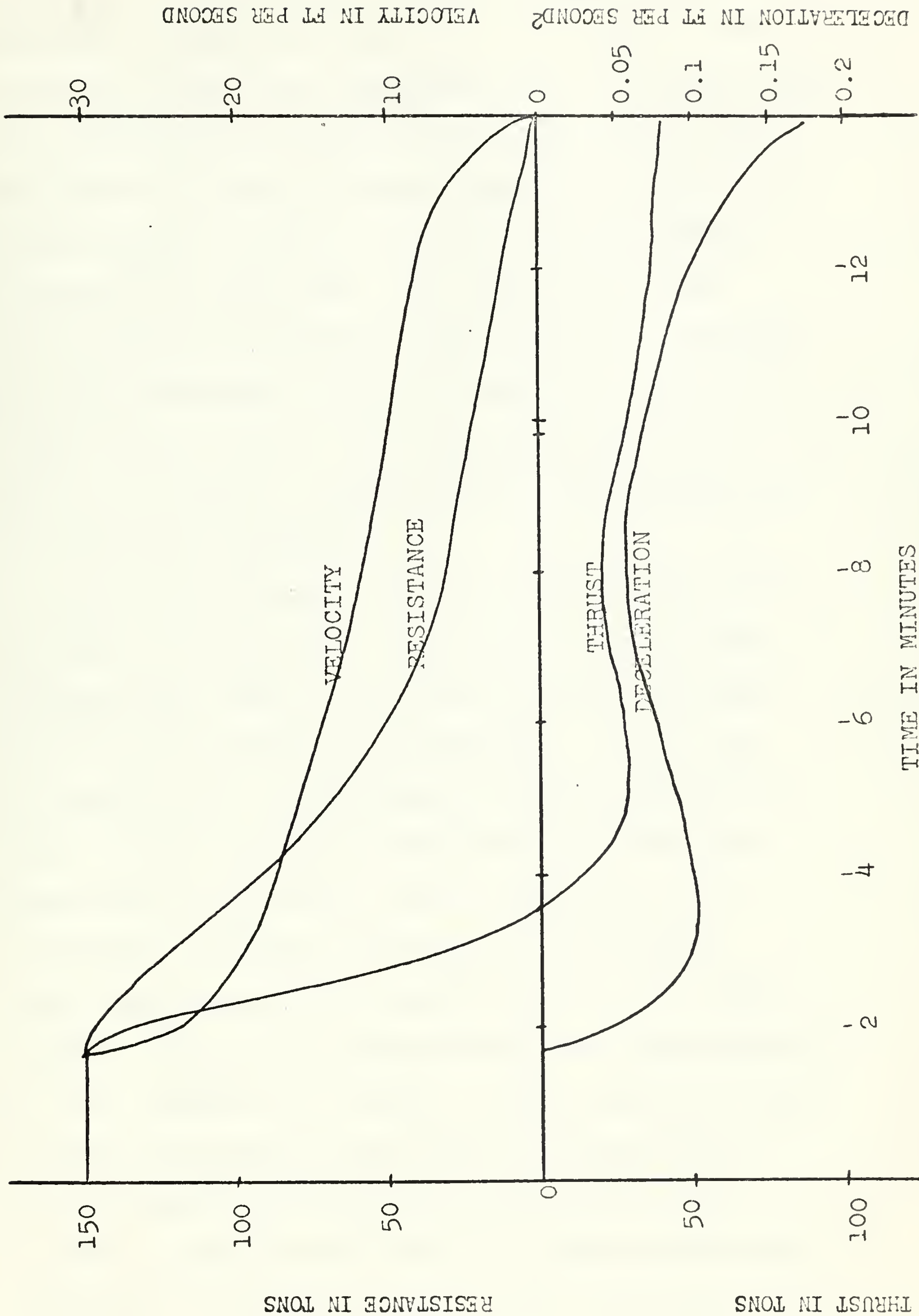
VELOCITY IN FT PER SECOND
DECELERATION IN FT PER SECOND²
30
20
10
0
0.05
0.1
0.15
0.2
VELOCITY
RESISTANCE
THRUST
DECELERATION
2
4
6
8
10
12
TIME IN MINUTES
150
100
50
0
50
100
RESISTANCE IN TONS
THRUST IN TONS

FIGURE 1

dynamics of stopping for a typical tanker without auxiliary braking devices. The relationship of the thrust T and the hull resistance R can be clearly seen. The propeller thrust does not become effective until the vessel is nearly stopped. Increasing astern power is not going to reduce the head reach appreciably because its effect comes into play after 75-80% of the total head reach has been traveled. Therefore the hull resistance R must be increased or auxiliary brake devices added if the stopping time and head reach are to be reduced. Increasing the hull resistance is obviously not the answer.

At first thought, the impact of such head reach may not seem important. However, in light of the International Rules of the Road[3] the large vessel runs into some tenuous circumstances. First, the well known rule of proceeding in limited visibility, that a vessel can make a speed such that her total head reach at a complete stop is one-half of the visibility. Secondly the rules require special lights and day-marks for vessels aground, not under command, fishing and sailing vessels;of such character as to be visible for 2 miles. A ship 1,000 foot long requiring 2-3 miles to stop from cruising speed is at a decided disadvantage when running into foul weather or upon another vessel of limited maneuvaring capability. A further impediment for the large vessel

is her inability to steer with the propeller turning astern in coming to a stop.

Investigations into the augmentation of brake devices has been thoroughly reported by only one source.[1] The device tested was a flap arrangement which could be extended from both sides of the ship into the slip stream for braking action and retracted for cruising. The model, a Todd Series 60 with block coefficient of 0.80, required a projected underwater flap area of 13-18% of the midship section area to reduce the head reach from initial speeds in the range of 8 to 16 knots by about 60%. The flap with 18% midship area was a solid wall flap which was later perforated to provide a projected area of 13% midship section area. The perforated flap gave a slightly better performance than the solid wall. These flaps were mounted in the forward part of the ship.

INTRODUCTION

DEVELOPMENT

It was felt that the utilization of flaps as a braking device could be improved upon. The flap as a practical device has some serious drawbacks. The first is that as a mechanical device it requires maintenance and overhaul to ensure its performance. It produces loads and moments which result in unnatural stress concentrations in terms of present day ship construction. Failure of one flap could result in loss of control of the directional stability of the ship.

The momentum reversing brake or impulse brake on the other hand, can be built into the structure of the ship, can have no mechanical parts except for an inlet valving arrangement, and could be up to 2 times as effective as a flap for the same projected area.

Both the flap and the impulse brake work best at high speeds and it is their purpose to slow the ship down quickly to that speed where the propeller thrust becomes effective in bringing the vessel to a dead stop.

The impulse device is well known for its performance at high Froude number flows (>100) in water turbines for hydroelectric power plants[5] and in brakes for rocket sleds.[9] Data was not available for the performance of the device at low Froude number flows (<2).

A series of tests at low Froude number flows of impulse devices (scoops) and flaps were in order to determine their performance characteristics.

PROCEDURE

The scoops and flaps were mounted on a test bed consisting of a flat plate with a pin-jointed parallelogram mounting to a frame such that the plate was free to move in plane. See figure 2 for a sketch of the apparatus. The plate and frame were suspended in the M.I.T. Propeller Tunnel with the flow parallel to the plane of the plate. The leading edge of the plate was sharpened and the plate itself was polished initially. The plate was prevented from swinging by an attachment to a Bytrex Model BC-50 load cell. This gave a direct reading of any drag force on the plate. The load cell was connected to a Sanborn Model 140, strain gauge indicator and recorder. The system was calibrated by using suspended weights.

The propeller tunnel was run with an open surface. The flow velocity was measured using the tunnel static-pitot tube connected to a water manometer. A water manometer was required due to the low velocities in the test section. The velocities in the test section were limited to sub-critical and for the flow depth were a maximum of about 4.6 FPS. The test section was brought supercritical upon occasion but meaningful data was not obtained as the flow speed and free surface were unsteady and the pitot tube and side walls caused an interferring wave pattern.

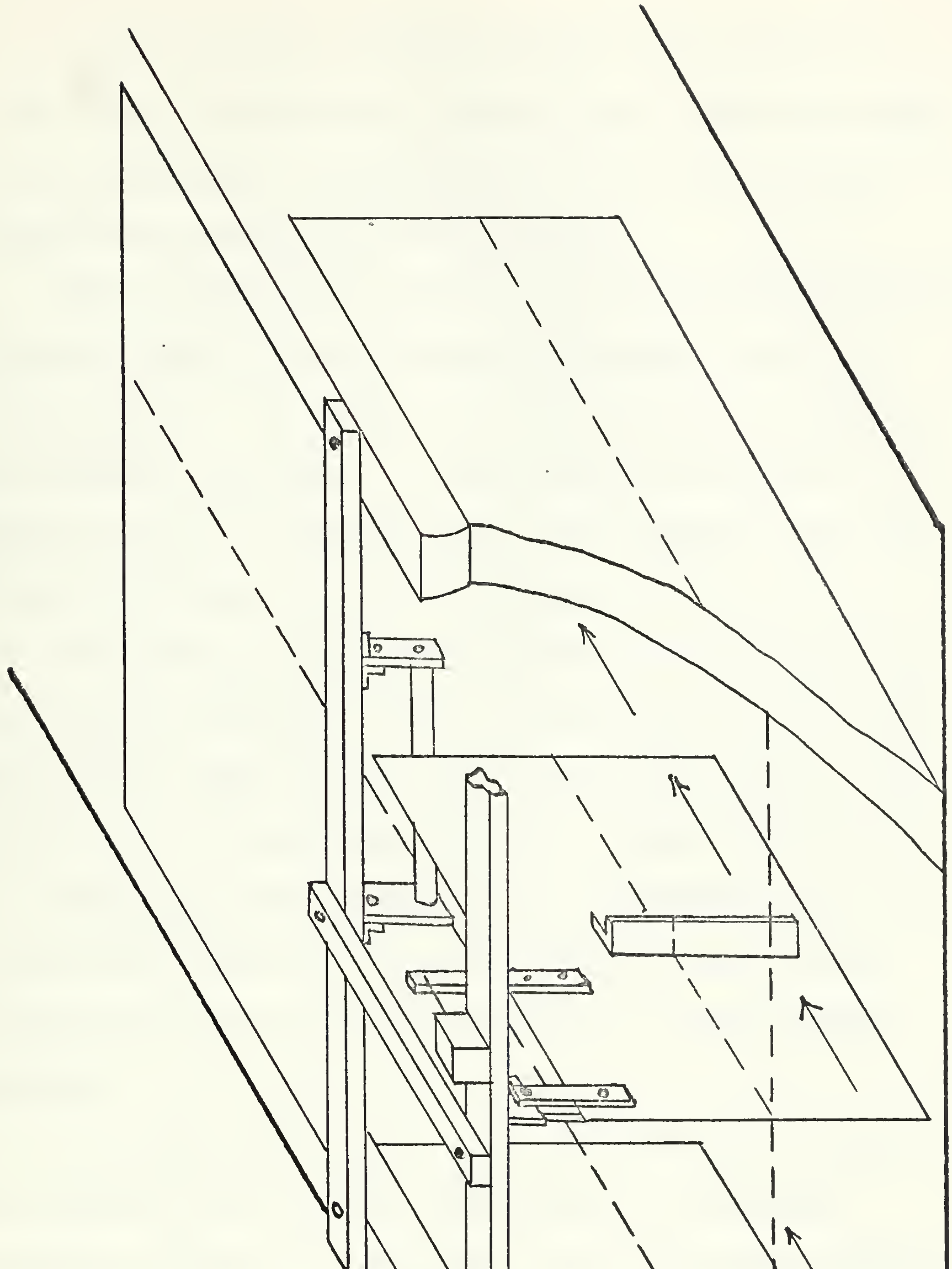

FIGURE 2.

A pair of flaps with underwater dimensions of 1"x4" were tested first. These flaps were to be used for comparison of performance of the scoops and to check the accuracy of the test rig.

Two scoops were tested. One was 6 inches high by 2 inches wide and the other 2 3/4 inches high by 2 inches wide. The scoops were geometrically similiar in heighth and depth. A side plate was mounted to keep the flow two dimensional. See figure 9 for the cross sectional view of the scoop.

The scoops were tested in the ram position, that is with the bottom of the scoop directly in the flow. Both scoops were tested as free and the smaller scoop with a free stream baffle plate as shown in figure 9.

Data runs were made in each configuration. Points were taken at intervals during trials with increasing and decreasing velocity to check repeatability.

Runs made with the scoops had a flap attached on the opposite side of the test plate to minimize sidewise torque on the plate. The resistance reading was corrected for the effect of the flap.

A prediction of the performance of the smaller scoop with the free stream plate using the Bernoulli equation as modified by the continuity equation is detailed in Appendix II.

A mathematical model of the deceleration of a

ship was developed from the equation of motion. The brake parameters developed from the experimental data along with the known parameters of an actual ship were introduced into the math model to give an order of magnitude analysis of the deceleration. The model is detailed in Appendix III.

RESULTS

TESTING PROGRAM

The test results for the flaps and various configurations of the scoops are plotted in the form of a resistance coefficient C_R based on $C_R = \frac{2R}{\rho A V^2}$, versus Froude number of the flow. Where

V is the free stream velocity, R is the drag, and A the area perpendicular to the free-stream. The characteristic length in the Froude number is the still water immersed depth of the flap and the heighth of the opening of the free scoop.

Figure 3 shows the resistance data of flaps. Figure 4 shows the data for the six inch scoop with ram flow and no free stream plate. Figure 5 is the data for the 2 3/4" scoop again with ram flow and no free stream plate. Figure 6 is the data for the 2 3/4" scoop with ram flow and a free stream plate inclined 10° to the horizontal. Figure 7 is a composite plot of all the test data along with the theoretical prediction of the small scoop with a free stream plate as developed in Appendix II.

Figure 10 shows the small scoop at a flow of $F_r = 1.03$ and figure 11 shows the small scoop at a flow of $F_r = 1.25$.

RESISTANCE DATA FOR 0.25 ASPECT RATIO FLAPS

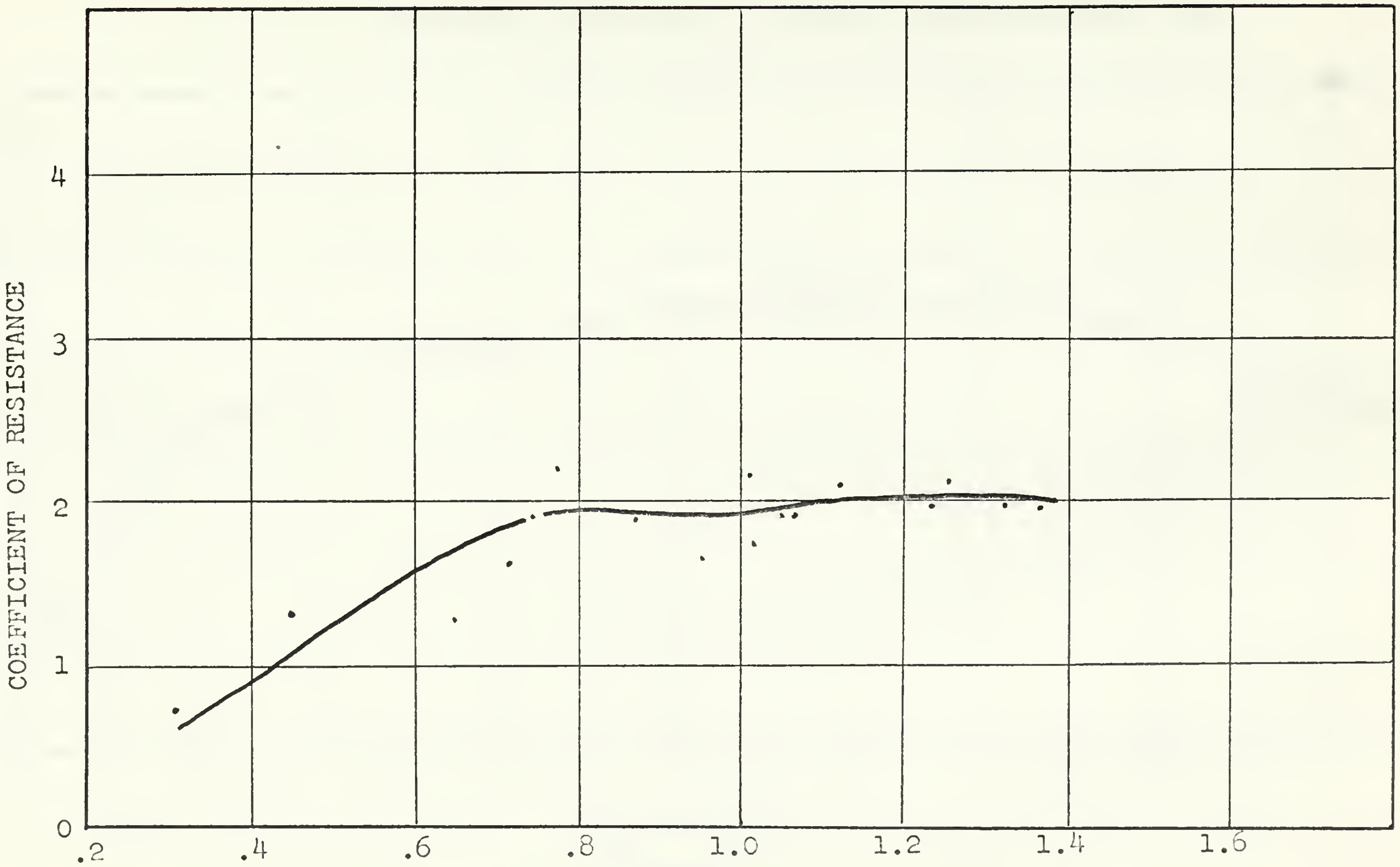

FIGURE 3.

RESISTANCE DATA FOR 1.0 ASPECT RATIO SCOOP W/O PLATE

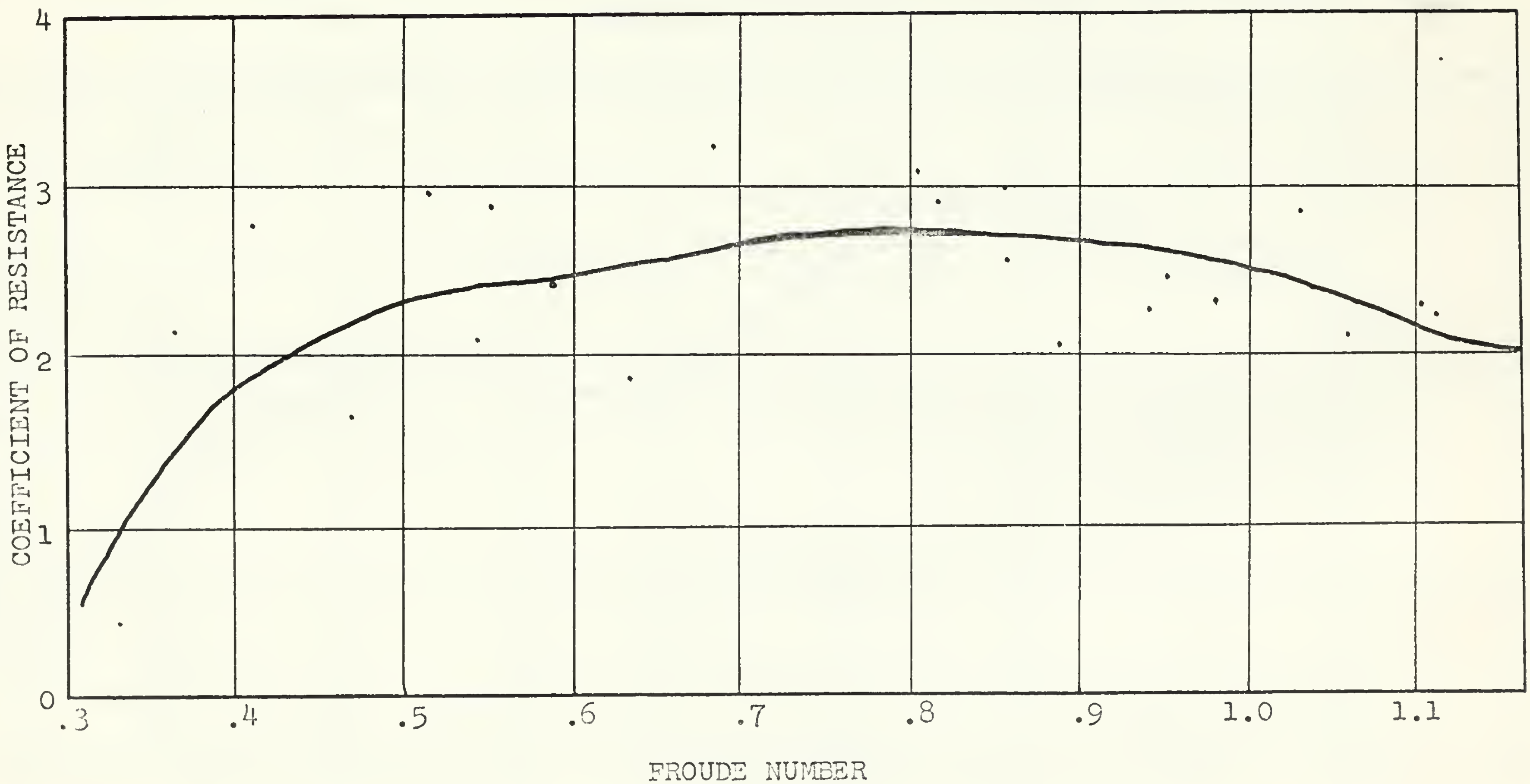

FIGURE 4.

RESISTANCE DATA FOR 2.0 ASPECT RATIO SCOOP W/O PLATE

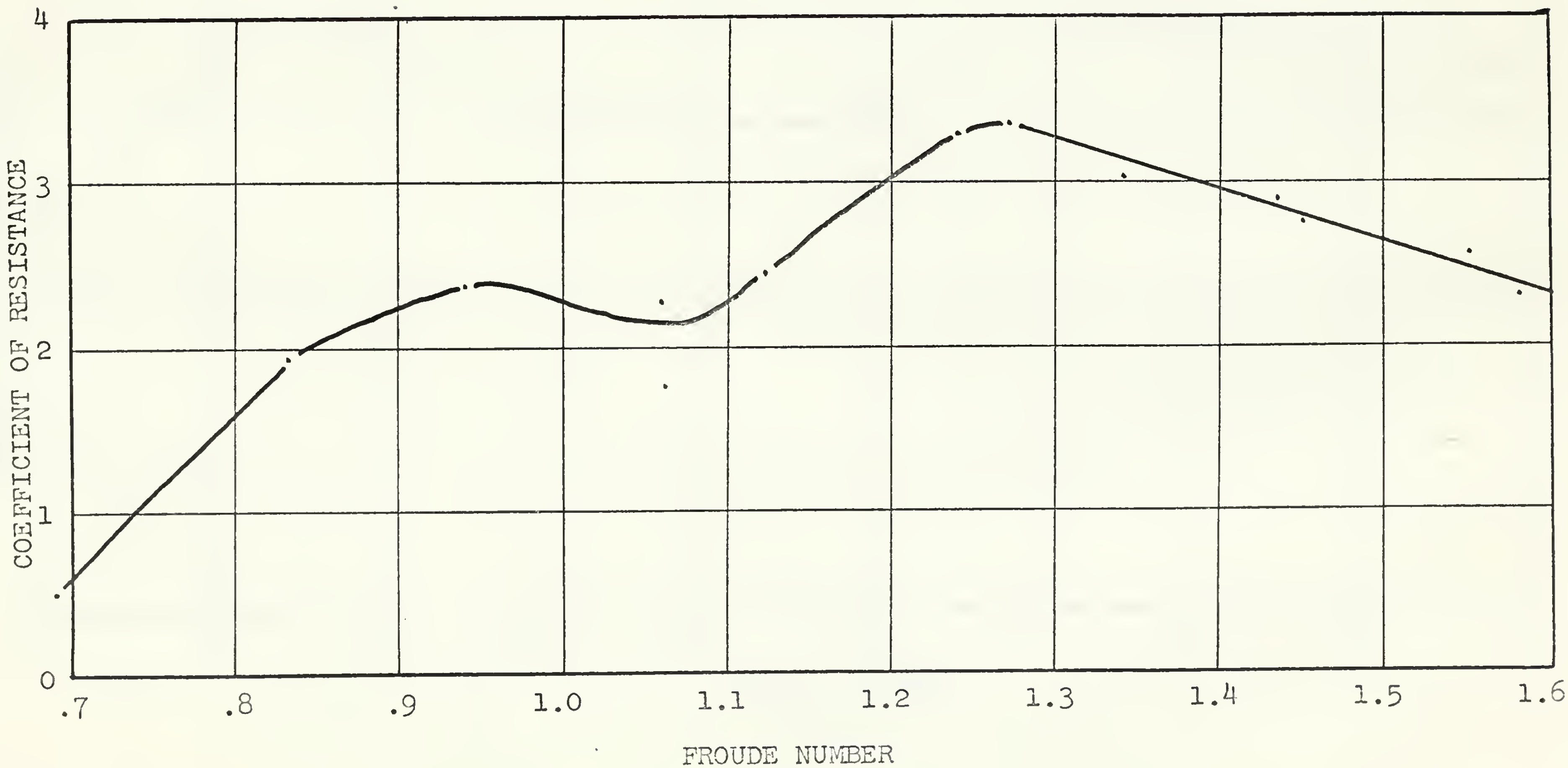

FIGURE 5.

RESISTANCE DATA FOR 2.0 ASPECT RATIO SCOOP WITH PLATE

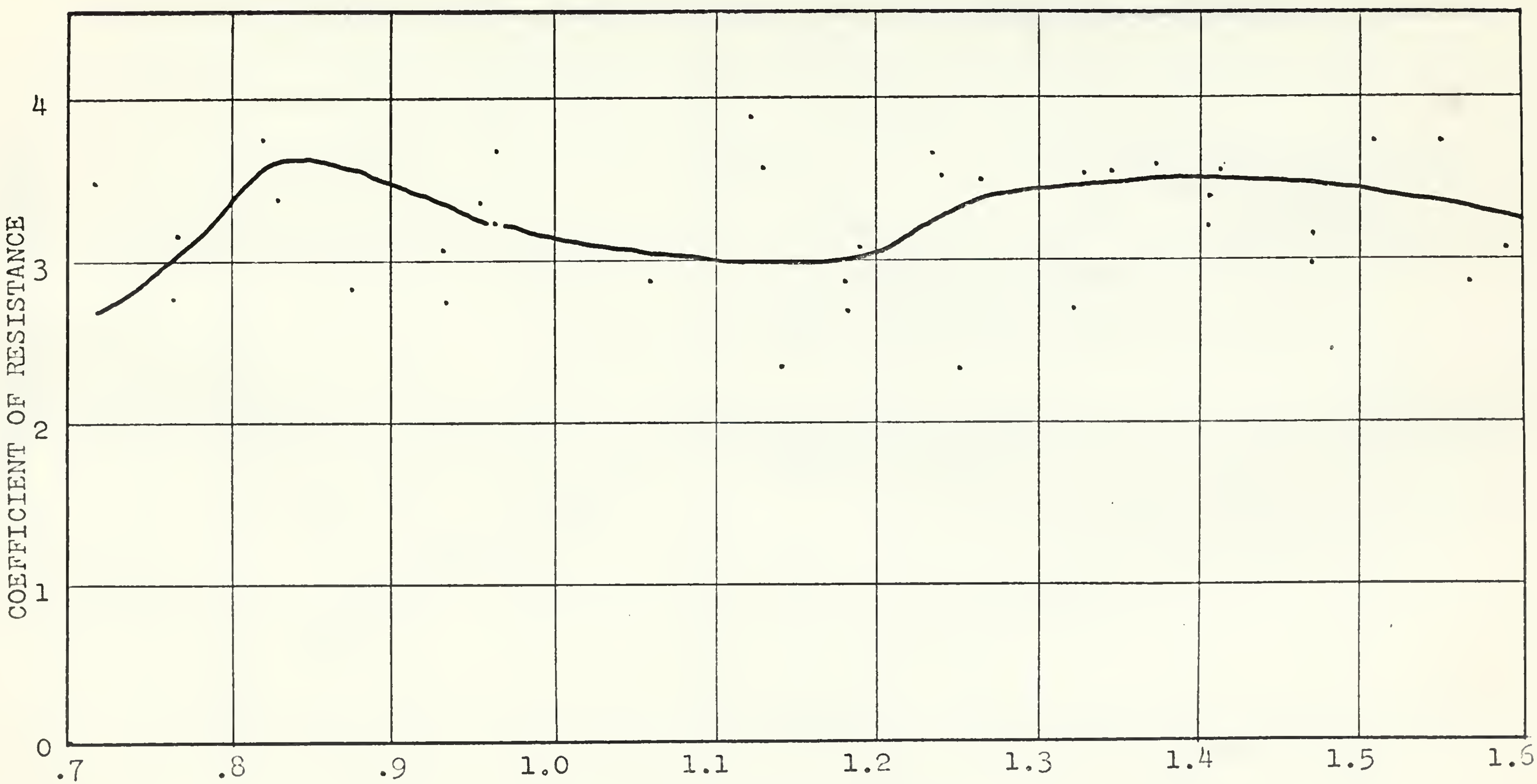

FIGURE 6.

COMPOSITE PLOT OF RESISTANCE DATA AND PREDICTED PERFORMANCE

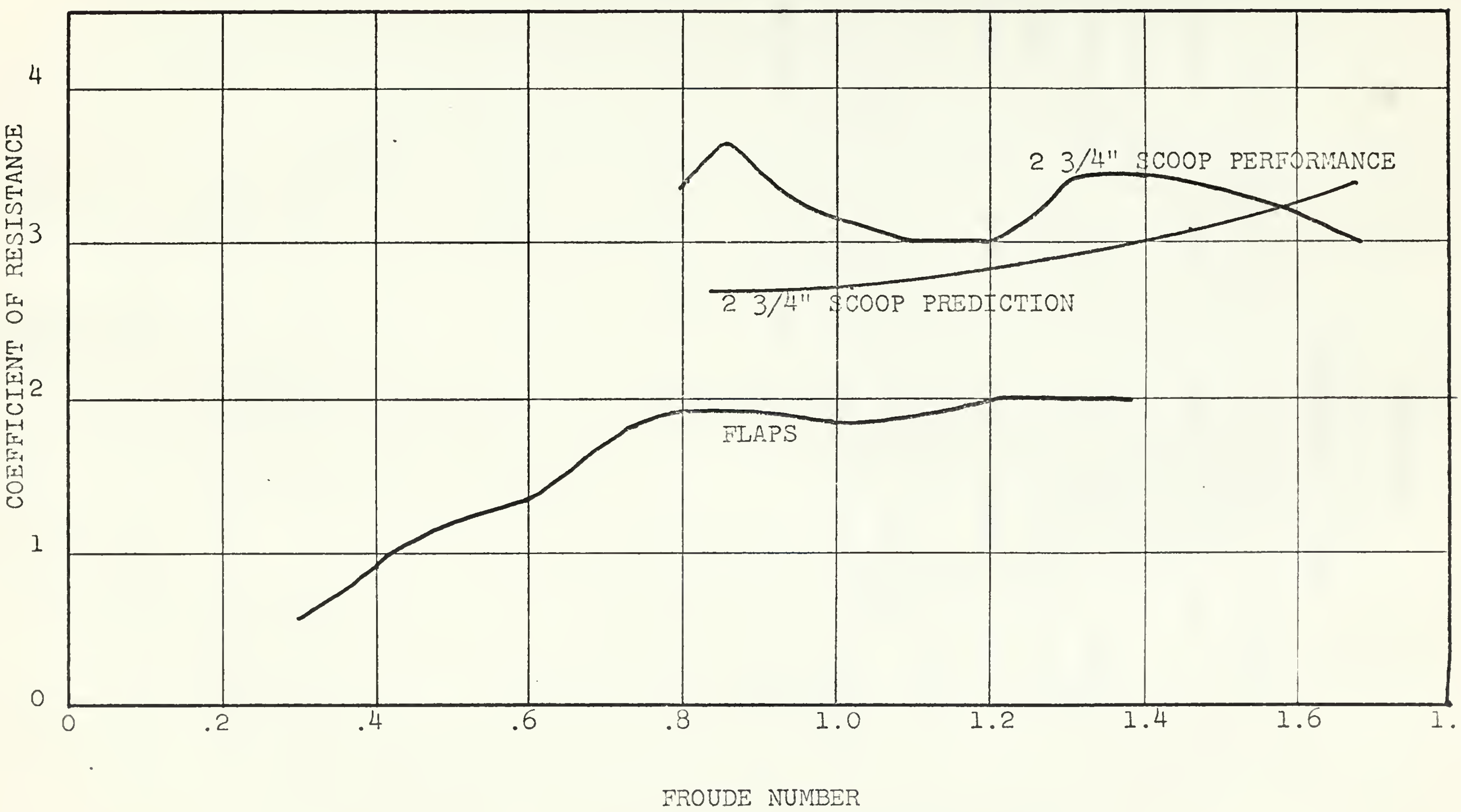

FIGURE 7.

RESULTS

MATHEMATICAL MODEL

The results of using the known parameters of an actual ship of 130,000 DWT (152,000 T. displacement) in the mathematical model of deceleration are given in the form of a plot in figure 8. The plot is of this particular ship slowing down from its' service speed of 15.5 kts. to 5 kts. The plot shows the relationship of braking area to the distance travelled. The model does not take into account the reverse thrust of the propeller.

HEAD REACH REDUCTION VERSUS BRAKE AREA

FOR 130000 DWT SHIP SLOWING DOWN FROM 15.5 TO 5 KNOTS

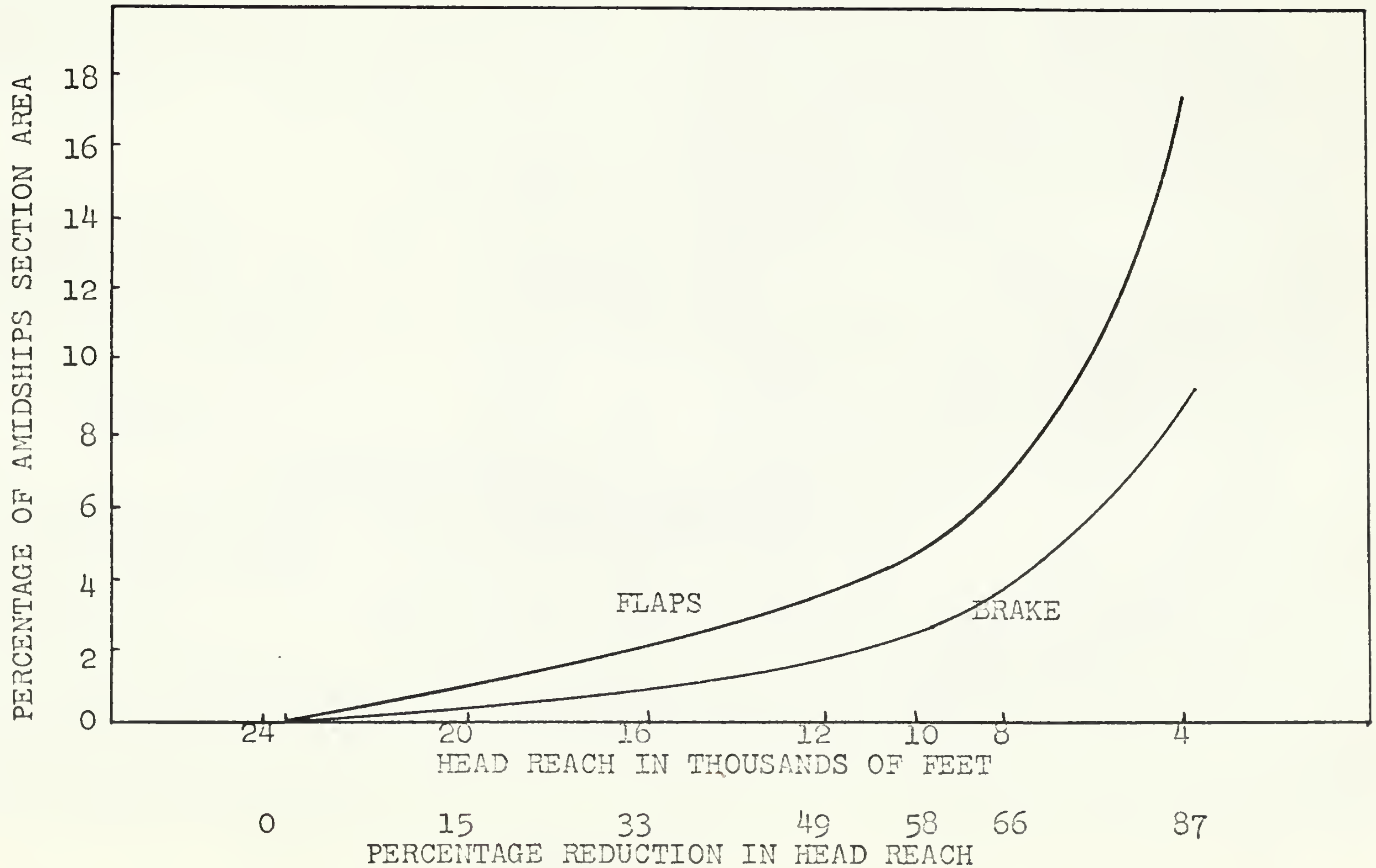

FIGURE 8.

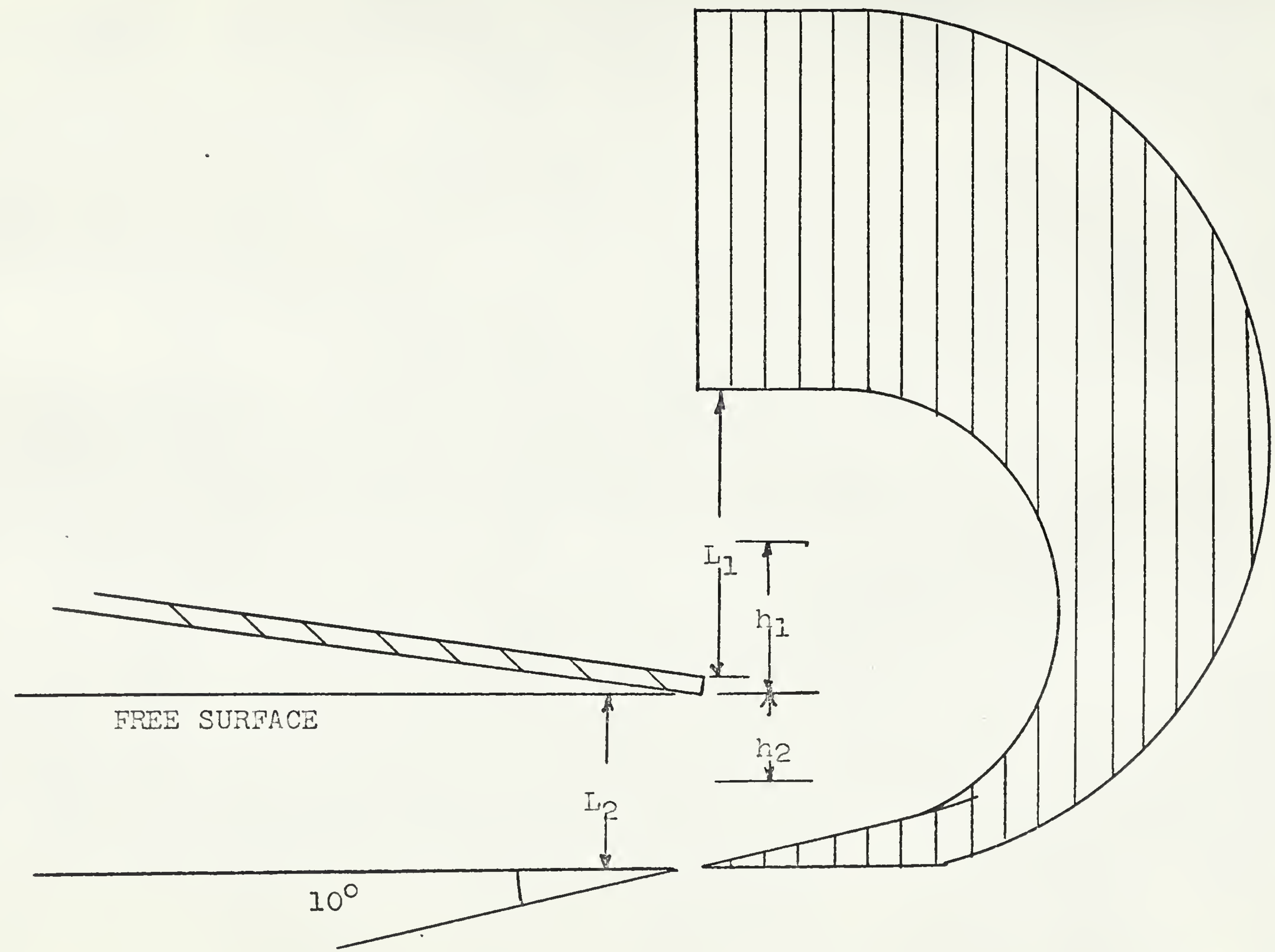

FIGURE 9.

FIGURE 10.

FIGURE 11.

DISCUSSION OF RESULTS

Before covering the results of the tunnel testing, the matter of aspect ratio of the underwater area should be covered. In making the models for the test, it was felt that they should approach the configurations of the actual devices. The flaps should be long and narrow and the scoop inlets short and wide. The matter of blockage in the tunnel had to be considered also. The final configuration was to have an aspect ratio of 0.25 for the flaps and 1.0 and 2.0 for the 6 inch and 2 3/4 inch scoops respectively. The aspect ratio so determined was based on width over heighth. The assumptions made in regards to the actual conditions were that the flaps would extend from the side of the ship and should not protrude too far. The brake inlet would probably be on the bottom of the vessel so as to maintain some effectiveness at different drafts. Therefore, the inlet if it is the ram type, should not extend very far below the bottom of the ship.

The results obtained with the flaps compared well with other investigations[8] and nothing really significant can be said except it provided confidence in the instrumentation.

The plot obtained of both scoops without the free stream plate showed a rising characteristic to a peak and then falling off after. The degradation of

of performance was easy to see during the tests as the discharge from the scoop would cascade down and disturb the flow coming into the scoop. The inflow disturbance was in the form of large air bubbles. The flow through the scoop would become unsteady after a period of time with the slugs of air causing feedback in the scoop performance and the flow would become pulsating. The interrupting slug of air can be clearly seen in figure 12.

It was this degradation which led to the installation of the free stream plate. The purpose, of which, was to eliminate or at least reduce, the interference of the inlet flow by the discharge. A horizontal plate was used first. However, the shape of the free stream surface, which changed with velocity, caused interferences at the leading edge of the plate and produced cavities at the trailing edge. The results obtained were sporadic and not repeatable. Inclining the free stream plate to an angle of about 10° above the horizontal, eliminated the problems of interference and cavitation. The discharge flow at the higher velocities running off the end and the side did tend to degrade the inlet flow but not to the same extent as the cases without a free stream plate.

The test data compared well with the predicted flow except at the lower flow velocities. Part of the problem comes from determining the suction load caused

FIGURE 12.

by partial ventilation of the after side of the scoop. The flow around the scoop was not completely clean and a wave was generated at the leading edge of the side plate. The resistance generated by the wave formation was not taken into account in the prediction.

The dip in all of the curves occurs about 3 FPS. A check of the flow at the pitot tube and the model, taking in account the blockage at the model, did not show any significant variances. An alternative explanation might be that it is caused by wave generation, although the effects of the primary wave generator, the flap, were removed from the determination. The last thought is that it might be caused from the bed plate, although resistance measurements on the plate alone were negligible.

The results of the math model analysis were optimistic. The figure of scoop area 2.5% of the amidships section area to reduce head reach by 60% appears low in light of the model tests of Jaeger and Joudain. But, they admitted that they could not correlate their model tests with the actual sea trials. In their correlation they attempted to account for the variance in braking forces using the Quasisteady Method by van Manen [10]. As the math model is of an actual ship, there is perhaps more credance in it. Tani[2] using a similiar math model with the same assumptions

but including provisions for the reversing thrust gave excellent results (less than 1% variance in head reach) when compared to sea trials. Attempts at arriving at a solution taking in all of the variable effects does not compare as well as the more simpler models[7]. In order not to cloud the issue, the propeller reversing thrust was not included in the braking force in this model. In actuality, of course, this would reduce the stopping distance and for the same stopping distance, would reduce the area of the brake required.

The use of a C_R of 3.9 was based on the experience of the testing. That is the brake in the ram configeration can very nearly approach a C_R of 4.0 where the discharge of the brake is just at the waterline and no additional head due to lift is to be overcome. This reinforces the assumption of a wide brake. Water turbine practice[5] puts the heighth of the bucket as about 5 to 6.5 times the heighth of the incoming fluid. The practice was not used here as the heighth ratio was 2.75 to 3.0. With the free scoop configuration (ie. not blocked and ducted) a 6 to 1 ratio would have given that much additional static head to be overcome.

The scoop arrangement tested was not the most advantageous, it is admitted. However, attempts to provide fairing around the scoop to study the impact

of suction drag led to blockage problems in the propeller tunnel and unsteady flow and most critically unsteady free surface.

CONCLUSIONS

The conclusions drawn from this thesis is that the momentum reversing brake is a significant improvement over that of flaps. The performance of the brake can be predicted by simple theory. Included in this, is the premise that the discharge should not interfere with the inlet flow. There is an additional dividend in suction drag due to cavitation or ventilation in the ram configuration. This coupled with a brake design of near perfect performance may actually provide a braking system whose coefficient of resistance would be greater than 4.0.

RECOMMENDATIONS

The ram configuration with its' attendent higher braking force has the mechanical problems associated with it as does the flap. Therefore it is recommended that further investigations of the flush inlet scoop be made in order to satisfy practical considerations. The scoop in this case would be ducted similar to a ship's condenser scoop.

The performance of the brake could be improved if the discharge flow could be made into a cavity below the water line. The addition of 10 feet of head on the inlet flow would yield an additional 25 FPS increase in discharge velocity which at 15 kts. ship speed could yield a brake C_R of nearly 6.0. The action of the flow at the interface of the cavity may be such as to negate the favorable head. The creation of a cavity or ventilated separation zone might lead to mechanical impracticality. But a study in the direction of improving the performance of the brake is recommended. Further improvement of the brake performance would lead to a proportionately smaller size. Smaller size would reduce its impace on reducting pay-load volume.

APPENDIX I

SYMBOL LIST

A	Cross sectional area perpendicular to the flow.
a	Acceleration.
B	Braking force in pounds or tons.
C_f	Coefficient of frictional resistance.
ΔC_f	Coefficient of frictional resistance correlation correction, constant = 0.0004.
C_r	Coefficient of residual resistance.
C_R	Apparent coefficient of resistance.
F	Force in pounds or tons.
g	Acceleration due to gravity 32.2 feet/second^2.
h	Average head in inches or feet.
L	Length in inches or feet.
M	Mass.
P	Pressure in pounds/feet^2.
P_a	Atmospheric pressure.
P_h	Hydrostatic Pressure.
R	Resistance in pounds or tons.
s	Wetted surface area.
S	Head reach.
t	Time in seconds.
T	Thrust in pounds.
V	Velocity in knots or feet/second^2.
ρ	Mass density.
η_P	Propulsive efficiency.

APPENDIX II

PREDICTION OF SCOOP PERFORMANCE

Part 1 Momentum change contribution

The flow through the scoop is governed by the Bernoulli equation as modified by the continuity equation. The scoop in question has a 1 inch high inlet and a 1 5/8 inch outlet as shown in cross-sectional view as shown in figure 9.

First some simplifying assumptions. This analysis will threshold when the incoming flow can support an average head of one-half of the discharge heighth. The velocity leaving the scoop is minimum at the top and maximum at the bottom, so that the average head seen is midway between. The slip stream plate has no effect with regard to the discharge head. Most of the discharge flow sweeps off to the side of the plate. Addressing the Bernoulli equation on the assumption the inlet and discharge areas are identical:

$$\frac{Pa}{\rho g} + \frac{V^2 in}{2g} + h_2 = \frac{V^2 out}{2g} + h_2 + h_1 + \frac{Pa}{\rho g}$$

Simplifying yields:

$$\frac{V^2 in}{2g} = \frac{V^2 out}{2g} + h_1$$

when $h_1 = 0.875$ inches, then $V_{in} = 2.16$ FPS.

The scoop has uniform width but the discharge heighth is different from the inlet heighth so that the discharge velocity will be modified by the continuity equation:

$$\rho_1 A_1 V_1 = \rho_2 A_2 V_2 .$$

Simplifying yields:

$$V_{actual} = V_{out} \frac{L_2}{L_1}$$

At the outset L, is equal to the full heighth of the discharge. As the inlet velocity increases the flow tends toward a more uniform cross section around the scoop and the flow discharge area decreases. After observing the flow, an estimate was made of the difference in the discharge area versus inlet velocity. The average head as seen by the incoming flow will increase slightly, but at the distances used here, the changes are negligible.

The braking force from an ideal impulse brake(11) is $B = \rho A V_{in}^2 + \rho A V_{out}^2$. When the fluid is incompressible, cross section areas are equal, and there is no head loss such that $V_{in} = V_{out}$; then $B = 2\rho A V_{in}^2$. the ideal coefficient of resistance is $C_R = \frac{2B}{\rho A V_{in}^2}$.

The ideal C_R with a brake with no losses is equal to 4.0. The minimum C_R would be 2.0 comparable to that of flaps.

SAMPLE CALCULATIONS

$$\frac{V_{in}^2}{2g} = \frac{h_1''}{12} + \frac{V_2^2}{2g} \qquad h_1'' = 0.875 \text{ inch.}$$

$$V_{in}^2 = 5.36 h_1'' + V_2^2 = 4.69 + V_2^2$$

$V_{in} = 2.16$ FPS Threshold.

$$V_{out} = \frac{L_2 V_2}{L_1} \qquad L_2 = 1.0 \text{ inch}$$

$$C_R = \frac{2(\rho A V_{in}^2 + \rho A V_{out}^2)}{\rho A V_{in}^2}$$

$$C_R = 2\left(\frac{V_{in}^2 + V_{out}^2}{V_{in}^2}\right)$$

$$F_r = \frac{V}{\sqrt{gL}} = \frac{V}{\sqrt{32.2 \times \frac{2.75}{12}}} = \frac{V}{\sqrt{7.36}} = \frac{V}{2.72}$$

PLOT DATA

V_{in}	V_{in}^2	V_2^2	V_2	L_2	V_{out}	V_{out}^2	C_R	F_r
2.2	4.84	0.15	0.39	1.62	0.24	0.057	2.02	0.81
2.5	6.25	1.56	1.25	1.50	0.83	0.69	2.22	0.92
2.8	7.84	3.15	1.78	1.50	1.19	1.42	2.36	1.03
3.1	9.61	4.92	2.22	1.38	1.61	2.59	2.54	1.14
3.4	11.56	6.87	2.62	1.38	1.90	3.61	2.63	1.25
3.7	13.69	9.00	3.00	1.25	2.40	5.76	2.84	1.36
4.0	16.00	11.31	3.37	1.25	2.70	7.29	2.91	1.47
4.3	18.49	13.80	3.72	1.13	3.29	10.80	3.19	1.58
4.6	21.16	16.47	4.06	1.13	3.6	12.96	3.23	1.69

Part II Prediction of suction drag

With the scoop in the ram position, the after part will ventilate when the velocity is high enough. Because of the nature of the device, the dynamic head is used up in the momentum reversal. The only head remaining is the static head due to the difference in water level before and after the scoop. An approximation to the suction force can be made by the hydrostatic pressure acting on the projected ventilated area.

To find the threshold of complete ventilation, Bernoulli is again recalled:

$$\frac{V^2}{2g} = \frac{h''}{12} \qquad h'' = 1$$

$$V^2 = 5.36$$

$$V = 2.32 \text{ FPS}$$

Force on scoop in pounds $F = P_h A = \rho \frac{ghA}{2}$

$$F = 64.4 \times 1 = 2.68 \text{ lb/ft}^2 = 0.0186 \text{ lb/in}^2$$

$$C_R = \frac{2F}{\rho A V^2} = \frac{2 \times 0.0186 \times 144}{2 \times 2 \times V^2} = \frac{2.68}{V^2}$$

SAMPLE CALCULATIONS

V	V^2	C_R	F_r
2.32	5.36	0.49	0.85
2.40	5.76	0.464	0.88
2.6	6.76	.396	.956
2.8	7.84	.342	1.03
3.0	9.00	.298	1.1
3.2	10.21	.262	1.18
3.3	11.89	.246	1.21
3.4	11.56	.232	1.25
3.6	12.96	.206	1.325
3.8	14.44	.185	1.4
4.0	16.0	.167	1.47
4.2	17.64	.152	1.545
4.4	19.36	.138	1.62
4.6	21.16	.126	1.69
4.8	23.04	.116	1.765
6.4	40.96	.0655	2.34
6.6	43.46	.0616	2.43

APPENDIX III

DEVELOPMENT OF SHIP DECELERATION MATHEMATICAL MODEL

The math model of ship deceleration is developed from the equation of motion. There are some simplifying assumptions applied to the model. The mass of the ship system is the mass of the ship plus the added mass of the water moving with the ship. The added mass is assumed constant during the range of the slowing process to be modeled and is equal to 5% of the ship mass.(1) The total coefficient of resistance of the ship, C_t, remains constant through the range of speeds being used. There is no effective propeller thrust. The starting equation is:

$$M \frac{dy}{dt} = -F,$$

where:

M is 1.05 x mass of the ship,

$\frac{dy}{dt}$ is the deceleration of the ship,

and F is the resistance forces on the ship.

$$F = R + B,$$

where:

$$R = R_r + R_f + \Delta R_f$$

$$R_r = \frac{C_r \rho sV^2}{2}$$

$$R_f = \frac{C_f \rho sV^2}{2}$$

$$\Delta R_f = \frac{\Delta C_f \rho sV^2}{2}$$

and B is the augmented braking force on the ship.

Transforming the differential:

$$\frac{dy}{dt} = \frac{VdV}{dS} = \frac{1}{2}\frac{d(V^2)}{dS}.$$

Substituting gives:

$$\frac{M}{2}\frac{d(V^2)}{dS} = -F = -KV^2,$$

where:

K is the resistance function of V^2.

$$\frac{d(V^2)}{V^2} = \frac{-2K}{M} dS$$

$$\frac{dV^2}{V^2} = \frac{-2K}{M} dS$$

$$\ln V^2 = \frac{-2KS}{M} \quad C_1$$

With the initial conditions that at $t = 0$, $V = V_0$ and $S = 0$, then

$$C_1 = \ln (V_0{}^2).$$

$$\ln (V_0{}^2) - \ln (V^2) = \frac{2KS}{M}$$

$$B = \frac{C_R}{2}\rho AV^2$$

$$K = \frac{\rho}{2}\left[(C_r + C_f + \Delta C_f)\ s + C_R A\right]$$

Therefore the equation of motion yields finally to:

$$\ln (V_0{}^2) - \ln (V^2) = \frac{S\rho}{M}\left[(C_r + C_f + \Delta C_f)s \quad C_R A\right]$$

SAMPLE CALCULATIONS

A 130,000 DWT tanker whose parameters were known was selected for use in the model. The major parameters required are as follows:

$L = 865'$	$C_B = 0.80$	$A_x = 7600$ ft.^{2}
$B = 145'$	$L/B = 6.0$	SHP = 27000 HP
$H = 53''$	$B/H = 2.75$	$\Delta = 152{,}000$ tons
$C_x = 0.99$	$V_o = 15.5$ kts.	$\eta_p = 0.65$

From Series 60(6)

$$s/\nabla^{2/3} = 6.0$$

$$s = 6\ (34 \times 1.52 \times 10^5)^{2/3}$$

$$s = 1.83 \times 10^5 \text{ ft.}^2$$

$$EHP = 27000 \times 0.65 = 17500 \text{ HP}$$

$$R = \frac{EHP \times 326}{V} \qquad V \text{ in knots.}$$

$$R = \frac{17500 \times 326}{15.5} = 3.69 \times 10^5$$

$$R = \frac{C_t \rho s V^2}{2} \qquad C_t = C_f + \Delta C_f + C_r$$

$$C_t = \frac{2R}{\rho s V^2} \qquad V \text{ in ft/sec.}$$

$$C_t = \frac{2 \times 3.69 \times 10^5}{1.99 \times 1.83 \times 10^5 \times 26.2 \times 26.2}$$

$$C_t = 2.94 \times 10^{-3} \text{ for } 15.5 \text{ kts.}$$

Going to series 60, C_t remains essentially constant on the average through the speed range down to 5 kts. Going to the math model to study the head reach in slowing down

from 15.5 kts. to 5 kts. using no braking devices:

$$\ln(26.2^2) - \ln(8.45^2) = \frac{S}{M}(C_t s)$$

$$6.54 - 4.26 = \frac{S(1.99)(2.94 \times 10^{-3})(1.83 \times 10^5)(32.2)}{(1.05)(1.52 \times 10^5)(2.24 \times 10^3)}$$

$$S = \frac{2.28 \times 1.05 \times 1.52 \times 2.24 \times 10^5}{1.99 \times 2.94 \times 1.83 \times 3.22}$$

$$S = 23{,}600 \text{ ft.}$$

Reducing S to 10,000 ft. by the use of braking devices:

$$2.28 = \frac{10^4 \times 1.99 \times 32.2}{1.05 \times 1.3 \times 2.24 \times 10^8}\left[2.94 \times 1.83 \times 10^2 + C_R A\right]$$

$$\left[C_R A + 538\right] = \frac{2.28 \times 1.05 \times 1.52 \times 2.24 \times 10^8}{1.99 \times 3.22 \times 10^5}$$

$$C_R A + 538 = 1272$$

$$C_R A = 734$$

If $C_R = 2.0$ for flaps, then $A = 367$ ft.^{2}

If $C_R = 3.9$ for brakes, then $A = 188$ ft.^{2}

367 ft.^{2} = 4.85% of amidship section area.

188 ft.^{2} = 2.5% of amidship section area.

PLOT DATA

S	%A_x FLAP	BRAKE
23,600	0.0	0.0
21,000	0.45	0.23
18,000	1.12	0.575
15,000	2.04	1.05
12,000	3.44	1.77
10,000	4.85	2.50
8,000	6.95	3.57
6,000	10.40	5.35
4,000	17.42	8.95

APPENDIX IV

EXPERIMENTAL DATA

The resistance data was read directly from the instrumentation and the flow speeds determined from the water manometer by the following equation:

$\frac{\Delta h}{12} = \frac{V^2}{2g}$, where Δh is in inches of water.

$V = 5.36 \Delta h$

The resistance data from the tests involving the scoops had to be corrected for the effect of one flap attached to the opposite of the test bed. This correction was 1/2 of the resistance reading for the test of the flaps alone for the corresponding velocity. The resistance of the test bed alone was negligible.

The different tests results are reproduced in the form of tables.

FLAPS

Test 1 Calibration 1 div. = 0.552 lb.

h	V	V^2	div	lb	C_R	F_r
1.0	2.32	5.74	0.9	0.50	1.57	0.707
2.15	3.40	11.55	2.3	1.27	1.98	1.05
3.05	4.05	16.4	3.3	1.82	2.00	1.23
2.50	3.66	13.4	2.8	1.55	2.08	1.12
2.25	3.47	12.0	2.3	1.27	1.91	1.06
1.80	3.11	9.65	1.6	0.88	1.64	0.950
1.20	2.54	6.44	1.4	0.77	2.16	0.775

FLAPS

Test 2 Calibration 1 div. = 0.221 lb.

h	V	V^2	div	lb	C_R	F_r
0.2	1.0	1.0	0.2	0.04	0.7	0.305
.4	1.47	2.15	0.7	.155	1.3	.45
.85	2.11	4.45	1.4	.31	1.25	.643
1.0	2.41	5.8	2.3	.61	1.9	.735
1.5	2.84	8.05	3.7	.82	1.83	.866
2.05	3.32	11.0	4.7	1.04	1.7	1.01
2.2	3.44	12.8	6.9	1.52	2.14	1.05
3.5	4.34	18.8	9.3	2.06	1.97	1.32
3.7	4.45	19.8	4.6	2.12	1.93	1.36
3.1	4.08	16.64	8.7	1.92	2.08	1.25
2.2	3.44	12.8	5.1	1.13	1.59	1.05
1.0	2.41	5.8	2.9	0.61	1.89	.735

6 INCH SCOOP W/O FREE STREAM PLATE

Test 1 Calibration 1 div. = 0.221 lb.

h	V	V^2	div	lb	lb_{corr}	C_R	F_r
0.35	1.37	2.56	0.4	.09	0.028	0.42	.331
.60	1.79	3.2	1.0	.22	0.11	1.32	.445
.90	2.2	4.84	2.4	.53	0.36	2.86	.55
1.20	2.54	6.44	3.1	.69	0.31	1.85	.632
1.8	3.11	9.66	4.5	.99	0.56	2.22	.775
2.2	3.44	11.8	7.1	1.57	0.91	2.96	.856
2.7	3.82	14.58	8.2	1.77	0.92	2.42	.95
3.2	4.14	17.1	10.0	2.21	1.26	2.83	1.03
3.6	4.40	19.3	10.0	2.21	1.16	2.30	1.10
3.4	4.27	18.2	9.2	2.00	1.00	2.11	1.06
2.9	3.94	15.5	8.2	1.77	.92	2.28	.98
2.35	3.56	12.65	7.0	1.55	.68	2.06	.886
2.2	3.44	11.8	6.5	1.44	.78	2.54	.856
1.65	2.98	8.88	3.4	.75	.34	1.47	.742
1.2	2.54	6.44	3.0	.67	.31	1.85	.632
0.65	1.88	3.54	1.2	.27	.15	1.63	.468
0.35	1.37	2.56	0.4	.09	.028	0.42	.331

6 INCH SCOOP W/O FREE STREAM PLATE

Test 2 Calibration 1 div. = 0.95 lb.

h	V	V^2	div	lb	lb_{corr}	C_R	F_r
0.4	1.46	2.13	0.2	0.19	0.12	2.16	.364
0.8	2.07	4.28	0.5	0.475	0.33	2.96	.516
1.9	3.23	10.4	1.4	1.33	.83	3.06	.805
3.7	4.44	19.6	2.3	2.19	1.14	2.23	1.11
2.7	3.81	14.5	1.8	1.71	.86	2.28	.94
2.0	3.28	10.75	1.5	1.42	.82	2.93	.816
1.4	2.74	7.5	1.1	1.04	.63	3.22	.682
0.5	1.64	2.69	0.3	0.285	0.19	2.71	.409

2 3/4" SCOOP W/O FREE STREAM PLATE

Test 1 Calibration 1 div. = 0.240 lb.

h	V	V^2	div	lb	lb_{corr}	C_R	F_r
0.4	1.47	2.16	0.4	0.095	0.018	0.6	0.54
0.95	2.26	5.10	1.5	0.36	0.13	1.91	0.83
1.55	2.89	8.32	2.6	0.61	0.20	1.72	1.06
2.20	3.44	11.80	5.0	1.20	0.55	3.39	1.27
2.80	3.88	15.00	6.0	1.44	0.59	2.83	1.43
3.30	4.21	17.70	6.8	1.63	0.63	2.55	1.55
3.75	4.50	20.25	7.2	1.71	0.64	2.49	1.66
3.45	4.30	18.50	6.7	1.59	0.59	2.30	1.58
2.90	3.94	15.50	6.1	1.46	0.59	2.75	1.45
2.45	3.63	13.20	5.5	1.33	0.56	3.02	1.34
2.10	3.36	11.30	4.4	1.07	0.52	3.30	1.24
1.60	2.89	8.32	2.8	0.77	0.26	2.24	1.06
1.20	2.54	6.41	2.5	0.59	0.21	2.35	0.94
0.65	1.87	3.50	0.6	0.14	0.025	0.50	0.69
0.25	1.16	1.34	0.4	0.095	0.025	1.30	0.42

2 3/4" SCOOP WITH FREE STREAM PLATE

Test 1 Calibration 1 div. = 0.461 lb.

h	V	V^2	div	lb	lb_{corr}	C_R	F_r
0.2	1.03	1.07	0.2	0.09	0.03	2.02	0.38
2.7	3.82	14.50	3.3	1.52	0.67	3.32	1.405
2.45	3.62	13.15	3.1	1.43	0.65	3.54	1.33
2.15	3.39	11.50	2.1	0.97	0.37	2.32	1.25
1.80	3.11	9.65	1.6	0.74	0.31	2.31	1.14
0.95	2.26	5.10	0.9	0.42	0.24	3.39	0.83
1.90	3.20	10.20	2.0	0.92	0.45	3.18	1.18
2.40	3.59	12.89	2.8	1.28	0.48	2.68	1.32
2.70	3.82	14.50	3.2	1.48	0.64	3.18	1.405
3.10	4.09	16.64	3.8	1.76	0.86	3.72	1.51
4.0	4.62	21.42	4.4	2.03	0.93	3.12	1.7
6.3	5.82	33.80	7.2	3.32	1.92	4.09	2.14

2 3/4" SCOOP WITH FREE STREAM PLATE

Test 2 Calibration 1 div. = 0.221 lb.

h	V	V^2	div	lb	lb_{corr}	C_R	F_r
0.2	1.0	1.0	0.2	0.044	0.022	1.59	0.368
0.8	2.08	4.32	1.4	0.31	0.165	2.76	0.765
1.3	2.54	6.45	2.9	0.64	0.27	3.02	0.932
1.35	2.59	6.70	3.1	0.68	0.31	3.33	0.954
2.0	3.27	10.66	4.4	0.98	0.48	3.24	1.20
2.1	3.36	11.30	4.8	1.06	0.55	3.50	1.24
3.3	4.21	17.70	8.5	1.88	0.91	3.71	1.55
3.7	4.46	19.85	8.9	1.97	0.91	3.31	1.64
3.5	4.33	18.70	8.3	1.83	0.80	3.08	1.59
1.95	3.25	10.55	4.2	0.92	0.42	2.86	1.19
1.9	3.23	10.40	4.3	0.95	0.44	3.04	1.19
1.3	2.54	6.45	2.7	0.61	0.24	2.68	0.932
0.7	1.94	3.76	1.4	0.31	0.18	3.47	0.714
6.2	5.78	34.50	13.6	3.00	1.58	3.30	2.13

2 3/4" SCOOP WITH FREE STREAM PLATE

Test 3 Calibration 1 div. = 0.221 lb.

h	V	V^2	div	lb	lb_{corr}	C_R	F_r
0.15	0.9	0.81	0.2	0.044	0.022	1.96	0.331
0.30	1.27	1.61	0.5	0.11	0.05	2.12	0.467
0.40	1.46	2.13	0.8	0.177	0.101	3.41	0.537
0.70	1.94	3.76	1.0	0.221	0.096	1.55	0.714
0.80	2.08	4.32	1.5	0.332	0.187	3.12	0.765
0.90	2.20	4.84	1.9	0.42	0.25	3.73	0.809
1.00	2.32	5.36	2.2	0.487	0.297	3.98	0.853
1.40	2.63	6.91	3.3	0.73	0.35	3.65	0.966
1.75	3.07	9.41	4.1	0.906	0.481	3.58	1.13
2.10	3.36	11.25	5.3	1.17	0.57	3.64	1.235
2.2	3.44	11.80	5.8	1.285	0.585	3.48	1.265
2.5	3.66	13.40	6.5	1.44	0.66	3.54	1.345
2.75	3.84	14.70	7.1	1.57	0.72	3.52	1.411
3.75	4.51	20.40	8.5	1.88	0.83	2.93	1.66
3.40	4.27	18.20	7.8	1.728	0.72	2.85	1.57
2.60	3.74	13.95	6.8	1.51	0.69	3.56	1.375
2.20	3.44	11.80	5.8	1.285	0.57	3.48	1.265
1.70	3.05	9.29	4.2	0.93	0.50	3.88	1.121
1.40	2.63	6.91	3.1	0.686	0.306	3.21	0.966

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